

# Reduced Instrumentation Heat Transfer Testing of Model Turbine Blade Cooling Systems

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## Summary

Turbine cooling system engineers often evaluate designs using experiments that measure heat transfer rates and pressure drop. Engine Reynolds numbers are reproduced in large-scale test sections and the dimensionless heat transfer coefficients then relate directly to engine values. The experimental test strategy allows the designer to quantify the performance of new, for example more intricate, flow schemes. One common method entails the use of a model manufactured from a thermally insulating material which is subjected to a rapid change in inlet gas temperature. This paper is concerned with a method of processing these tests by only analysing the gas temperature changes. The approach offers data at lower resolution compared to conventional methods but can provide information in situations where full surface temperature changes can not be measured. The advantages and disadvantages of the method are discussed and results are compared to the data from a conventional analysis of liquid crystal coated models.

## Background and purpose of research

Several experimental methods have been used in the evaluation of turbine cooling system heat transfer performance by numerous researchers. The techniques used include the steady state conducting element method (Florschuetz et al., 1981) and the steady state uniform heat flux approaches (Hippensteele and Russell, 1984). A transient heat transfer method using a temperature sensitive paint was first applied to a turbine internal cooling system by Clifford et al. (1983). Since this early paper, there have been numerous examples of the technique to blade cooling research. Baughn et al. (1989) compared the transient to the uniform heat flux method. Recent transient method publications include Son et al (2000), Azad et al. (2000) and Gillespie et al. (2000). For a detailed description of the method, the reader is referred to a review of the technique given in Ireland and Jones (2000). The technique benefits from reasonably straightforward model manufacture and is able to provide high-resolution data from short duration experiments. Figure 1 shows a rig used at Oxford to measure rib cooling system performance. One advantage of the standard transient method is that local heat transfer coefficients are determined over any region of interest for which surface temperature data is available.

The analysis route covered in this paper offers data at much lower resolution, but, under some circumstances, the area-averaged data can be of great use. The reasons behind the development of the technique are listed below.

1. **Independence from surface temperature thermometry.** The technique does not require the surface temperature to be logged during the experiment. The result is that experiments can be performed for rigs for which optical access is not possible. This

Report Documentation Page				Form Approved OMB No. 0704-0188	
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1. REPORT DATE <b>00 MAR 2003</b>		2. REPORT TYPE <b>N/A</b>		3. DATES COVERED <b>-</b>	
4. TITLE AND SUBTITLE <b>Reduced Instrumentation Heat Transfer Testing of Model Turbine Blade Cooling Systems</b>				5a. CONTRACT NUMBER	
				5b. GRANT NUMBER	
				5c. PROGRAM ELEMENT NUMBER	
6. AUTHOR(S)				5d. PROJECT NUMBER	
				5e. TASK NUMBER	
				5f. WORK UNIT NUMBER	
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) <b>NATO Research and Technology Organisation BP 25, 7 Rue Ancelle, F-92201 Neuilly-Sue-Seine Cedex, France</b>				8. PERFORMING ORGANIZATION REPORT NUMBER	
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)				10. SPONSOR/MONITOR'S ACRONYM(S)	
				11. SPONSOR/MONITOR'S REPORT NUMBER(S)	
12. DISTRIBUTION/AVAILABILITY STATEMENT <b>Approved for public release, distribution unlimited</b>					
13. SUPPLEMENTARY NOTES <b>Also see ADM001490, presented at RTO Applied Vehicle Technology Panel (AVT) Symposium held in Leon, Norway on 7-11 May 2001, The original document contains color images.</b>					
14. ABSTRACT					
15. SUBJECT TERMS					
16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT <b>UU</b>	18. NUMBER OF PAGES <b>14</b>	19a. NAME OF RESPONSIBLE PERSON
a. REPORT <b>unclassified</b>	b. ABSTRACT <b>unclassified</b>	c. THIS PAGE <b>unclassified</b>			

can be useful in complex cooling systems for which it is impossible to gain optical access to the full heat transfer surfaces because of a limited number of cameras or potential sites for cameras. Often, the lighting requirements for one surface prevent another surface from being properly illuminated or viewed. The finite thickness of an encapsulated liquid crystal coating limits the maximum heat flux that can be measured in practice – see Ireland and Jones (2000). The new technique opens the way for higher heat flux experiments to be performed.

2. **Reduced manufacture time.** The tests analysed using the new procedure have used existing Perspex models designed for conventional transient, heat transfer experiments using liquid crystals. However, analysis of only gas temperature changes means that model requirements for optical access can be relaxed. This means that the cooling system model can be produced from material that allows more rapid manufacture than the Perspex. A softer plastic material such as a wax is easier to machine to the required scale size. The material does not need to be transparent and does not need to be polished as the heat transfer experiments rely on gas temperature measurements. The model is fitted with gas thermocouples, as before, though black ink and liquid crystal is not required.
3. **Reduced rigging time.** The model can be fitted to the same rig used for transient heat transfer testing. However, there is no need to position lights or to align cameras to illuminate and observe the liquid crystal. The rig set up time is considerably reduced and the need for a crystal calibration stage is also eliminated.

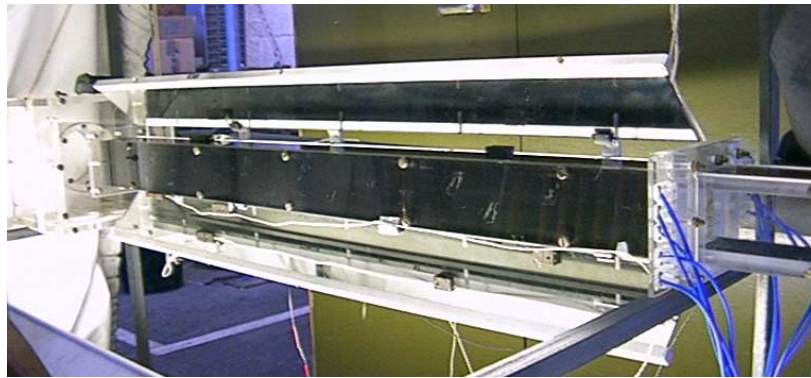


Figure 1 Apparatus used for rib heat transfer experiments.

### ***Application of gas temperature measurements in transient experiments***

The heat transfer coefficient distribution in a conventional transient heat transfer experiment is calculated from a procedure that employs a functional relationship between the local surface temperature change and the local heat transfer coefficient. Knowledge of the relevant gas temperature is required to determine the  $h$ , together with the model initial temperature, crystal calibration and thermal properties of the model material. The method requires the heat transfer driving gas temperature, specifically  $T_g$  in the equation (1) for local heat flux,  $q$ ,

$$q = h(T_g - T_s) \quad (1)$$

to be measured directly or derived from experimental data. A decision has to be made on what driving temperature to use for the calculation of heat transfer coefficient. In the simplest case, the  $h$ s are calculated based on an upstream driving temperature measured using one or more gas thermocouples. This strategy was used by Ireland and Jones (1985) to measure the heat transfer coefficient round an isolated pedestal. The local htc maps were based on the centreline temperature of the developed channel flow ahead of the pedestal. In longer passages, it is advantageous to calculate the  $h$ s based on a local fluid temperature. Clifford et al (1980) measured the duct centreline temperature through models of complex turbine cooling passages. In cooling systems with side bleed, eg Gillespie et al (2000) and Hwang et al (1999), it can be difficult to establish a representative heat transfer driving temperature.

After the htc data have been acquired, in subsequent component analysis, changes in mixed bulk temperature are directly related to total heat flows so  $h_s$  based on local mixed bulk temperatures can be especially useful. There are also good physical reasons why  $h$  based on a local temperature, such as local mixed bulk, are less sensitive to thermal boundary conditions. A method of calculating  $h_s$  based on mixed bulk temperature was introduced by Metzger and Larson (1986) for flow round a bend. A review of different strategies was included in Chyu et al. (1997) and later in Tsang et al. (2000). Gillespie et al. (1994) were able to use knowledge of the heat transfer coefficient based on the local mixed bulk temperature at the exit to a branched duct to process liquid crystal heat transfer experiments for which the entry mixed bulk temperature was unknown.

In the present paper, the surface temperature is not used to determine  $h$ . Instead, the gas temperature signals are processed in one of two ways. The choice depends on practical concerns such as how rapidly the air supply temperature can be switched and the time available for analysis. Both methods are described and compared below.

### **A. Initial temperature distribution method**

For experimental rigs that can achieve an abrupt change in the supply gas temperature, the turbine cooling system is analysed by extrapolating the measured gas temperatures back to the beginning of the experiment. At the start of the experiment, the model temperature is uniform and known. This allows an energy balance to be performed to determine the intervening, mean heat transfer coefficient.

The change in the gas temperature between planes is analysed to yield the number of transfer units (defined below) between the measurement planes. The NTU is based on the average heat transfer coefficient between the planes. In the absence of bleed flow to films, the mathematics is straightforward. For an initial wall temperature  $T_w$  and initial, mixed bulk gas temperature changes between two planes from  $T_{mb(i-1)}$  to  $T_{mbi}$ , then

$$NTU = \frac{hA}{mc_p} = \ln \left[ \frac{T_{mb(i-1)} - T_w}{T_{mbi} - T_w} \right] \quad (2)$$

where  $h$  is the average heat transfer coefficient between planes  $i-1$  and  $i$ ,  $A$  is the wetted area between the planes and  $m$  and  $c_p$  are the mass flow rate and specific heat respectively.

The initial gas temperature distribution needs to be accurately determined for the *initial temperature distribution method*. Attention needs to be paid to the model material thermal product<sup>1</sup> and to the gas temperature instrumentation bandwidth. One approach is to ensure that the internal surface temperature does not change significantly within the longer of (a) the flow establishment time or (b) the temperature transducer time constant. Figure 3 shows how long it takes the substrate temperature difference to reach 5% of its asymptotic value under circumstances in which the gas temperature has undergone a step change. The authors have used Perspex models that change temperature relatively rapidly. They found it necessary to extrapolate the gas temperature signals back to the start of the experiment.

The tests discussed below used a machined finish for which the surface average roughness was judged to correspond to a hydraulically smooth condition. It is worth noting, however, that, since there is no need to apply a liquid crystal coating, it is possible to machine the surface with a roughness at scale that is representative of a cast blade.

### **B. Gas temperature variation with time method**

The gas temperature data throughout the test can be compared to gas temperatures predicted from an analytical model of the experimental gas temperature rise. The latter analysis requires knowledge of the average  $h$  between planes to predict the temperature field so a

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<sup>1</sup>  $\sqrt{\rho k}$

regression procedure is used to determine the most likely  $h$ . The advantage of this approach is that it can be used for experiments where it is not possible to produce a sufficiently abrupt start. The analysis method is described in Tsang et al. (2000) and is related to the approach introduced by Wolfersdorf et al. (1997) who introduced an analytic model that enabled mixed bulk heat transfer coefficient to be determined from surface temperature. Similar strategies have been used to assess the performance of heat exchanger matrices- see Smith (1996).

In the work used in the evaluation of the new methods, the temperature at the exit to the ribbed section was compared to that predicted using a numerical model of the duct. The measured inlet temperature was used as an input, together with the starting model temperature and the measured mass flow rate. The duct is dividing into ten sections and the wall temperature and fluid temperature variations are calculated in an iterative procedure for a time matching the duration of the experiment. Regression was then used to determine the heat transfer coefficient that gave the best fit to the measured exit temperature.

Requirement	A. Initial temperature distribution method	B. Gas temperature variation with time method
Model initial temperature is uniform.	Required	Preferred. However, could be analysed if the variation is known.
Gas temperature distribution through the duct must be determined at the start of the experiment, before the surface temperature rise is significant.	Required	No. However, gas temperature sensor time constants must be small or known and accounted for.
The relationship between the temperatures measured at any plane and the mixed bulk temperature, $\phi$ , is known.	Required	Required
Thermal properties of substrate must be known.	Not required	Yes
Model material must be homogenous	No. Only needs to have uniform initial temp.	Yes. Otherwise analysis is very complicated.

Table 1 Summary of the requirements for the 2 methods.

### **Gas Temperature Instrumentation**

Different sensors can be used to measure the gas temperature. These include:-

- An adiabatic (nylon) mesh threaded with thermocouples. This approach is used at Oxford- see Tsang et al. (2000) and Mee et al. (2000). Figure 2 is an example of an arrangement used in the ribbed duct heat transfer research discussed below
- An adiabatic (nylon) mesh coated with liquid crystals then monitored with a video camera. This method can provide a detailed temperature profile although the disadvantage is that optical access is required. See Wang et al. (1998) and Mee et al. (2000) for examples of its application.
- An adiabatic (nylon) mesh threaded with platinum resistance thermometer(s). This approach produces a single signal for the temperature averaged along the length of the wire. A platinum wire threaded through the mesh has a resistance change proportional to the average temperature change of the wire.

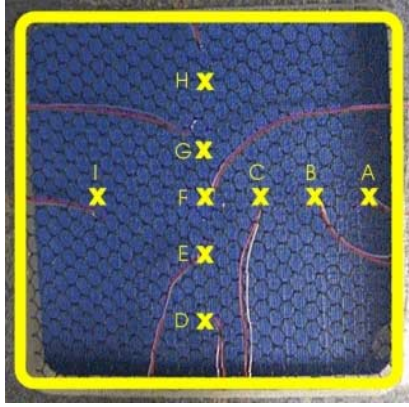


Figure 2 Thermocouples woven in to nylon mesh.

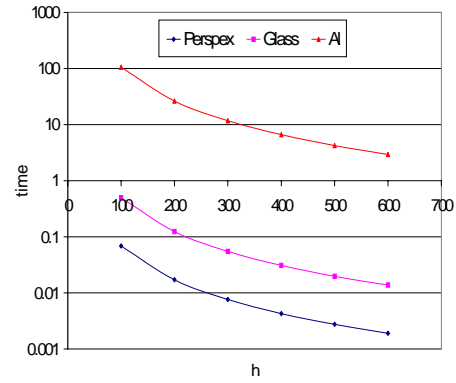


Figure 3 Time required for substrate dimensionless temperature to change 5% for different materials as a function of heat transfer coefficient.

Whichever way the temperature data is to be analysed, the experiment needs to give the local mixed bulk (or mixing cup) temperature at positions within the model. Evaluation of the mixed bulk temperature at any plane requires an integration of the temperature velocity product. In cases where it is not possible to measure the velocity distribution, it is proposed that the latter be estimated using knowledge of the total coolant flow at each plane and assumed velocity profiles. Dependent on rig size, the temperature distribution can be determined with lower uncertainty using several temperature sensors at each plane. The implication of uncertainty in the profiles on uncertainty in  $h$  is discussed in detail below. The analysis also leads to guidance on the minimum separation between the planes.

The link between the measured temperatures and the local mixed bulk temperature is considered by introducing a profile parameter,  $\phi$ , that relates the measured gas temperature, indicated here by  $T_{EXP}$ , to the true mixed bulk temperature,  $T_{mb}$ .

$$\phi_i = \frac{T_{mbi} - T_w}{T_{exp i} - T_w} \quad (3)$$

The subscript  $i$  refers to a number which identifies the measurement plane. Note that the measured temperature can be the average temperature, measured using for example by a PRT, or can be determined from multiple thermocouples.  $T_w$  is the model wall temperature.

For method (A), equation (2) requires the mixed bulk temperature to be measured and so the temperature profile factor can be used to express the mixed bulk temperatures in terms of the measured temperature. Equation (2) becomes

$$\frac{hA}{mc_p} = \ln \left[ \frac{\phi_{i-1}}{\phi_i} \cdot \frac{T_{exp(i-1)} - T_w}{T_{exp i} - T_w} \right] \quad (4)$$

### Evaluation of the methods

The above methods were used to determine the average heat transfer coefficient in a cooling passage in which ribs had been included to increase the heat transfer. The research was part of a program to evaluate the performance of turbulators with a larger size (relative to the passage hydraulic diameter) than those traditionally used in cooling passages. Some of the findings of the work are presented in Tsang et al. (2001). Data from the above technique are compared below to those evaluated using the conventional, transient liquid crystal method.

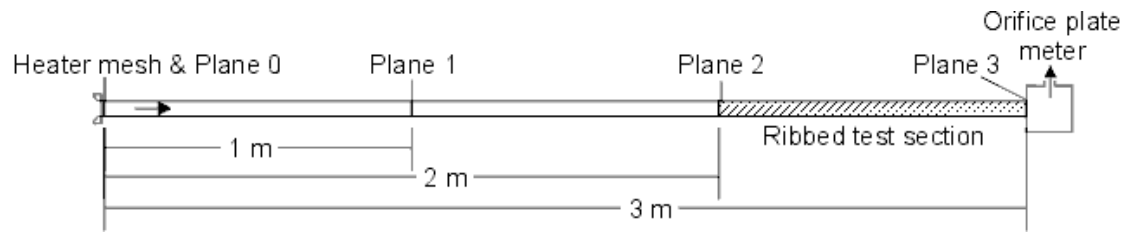


Figure 4 Schematic diagram of the passage used for rib heat transfer research.

Figure 4 shows the rig together with the positions of the gas temperature measurement planes. The Perspex rig has internal section with 50mm square sides. The air is drawn from atmosphere through the bell mouth intake and passes immediately through a fine wire, heater mesh supplied with AC power from a transformer. The airflow is initiated and adjusted to achieve the test Reynolds number and heat transfer starts when power is suddenly switched to the mesh heater. Gas temperature signals logged throughout a typical experiment are shown in Figure 5.

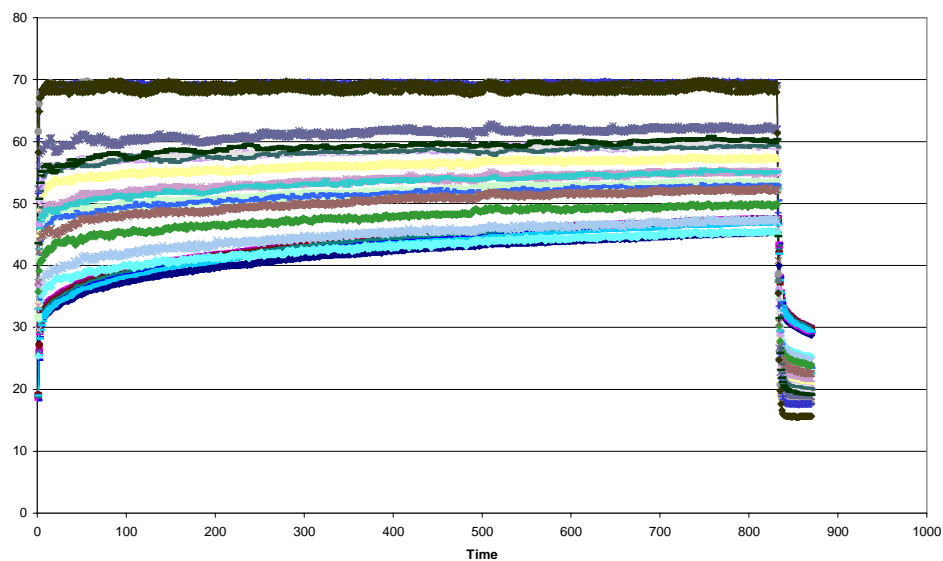


Figure 5 Logged gas temperatures for a typical experiment.

At each plane, the gas temperature was measured using fine ( $76\mu\text{m}$  wire) diameter Copper Constantan thermocouples threaded through an adiabatic mesh as described in Mee et al. (1999). The thermocouple time constant reduces as the flow speed increases. The former was calculated to be 50ms at the lowest Reynolds number tested. The longest flushing time was used to estimate the time necessary for the flow field to establish. Dividing 3m by the lowest velocity gives a flushing time of less than 1second. To minimise uncertainty in the start temperature, the gas temperatures analysed were determined by curve fitting to the signals between 1 and 10 seconds after the heat initiation, Figure 6. Figure 2 shows the thermocouple positions in each of the gas temperature measurement planes. All of the thermocouples were logged, but two different sets of thermocouples were used in the analysis.

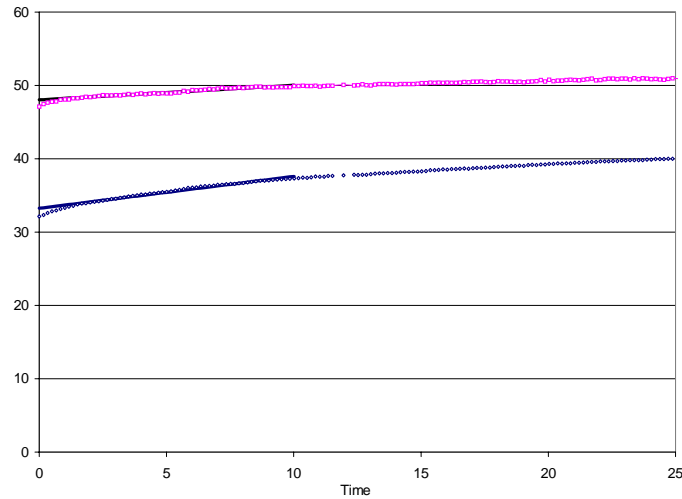


Figure 6 Extrapolation used to determine the initial gas temperatures.

### Temperature profile factors

The profile factor was evaluated using velocity measurements, Figure 7, made with the miniature probe described in Tsang et al. (2001). Since the flow through the smooth development section was turbulent, the velocity and temperature profiles will be similar by Reynolds analogy. This means that the distribution of the dimensionless velocity ( $u/u_{cl}$ ) is the

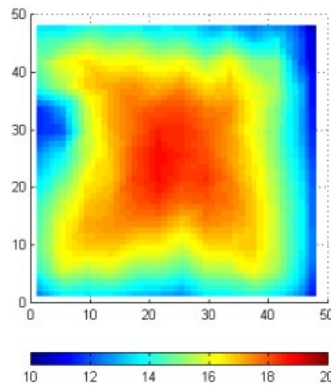


Figure 7 Velocity distribution at plane 2. This is downstream of the smooth development section and ahead of the ribbed walls.

same as the distribution of dimensionless temperature  $(T-T_w)/(T-T_{cl})$ . The mixed bulk temperature is defined as

$$T_{mb} = \frac{\int \rho u c_p T . dA}{\int \rho u c_p . dA} \quad (5)$$

which simplifies to

$$T_{mb} = \frac{\int u T . dA}{\int u . dA} \quad (6)$$

since the density and specific heat are uniform across the measurement plane. A little algebra leads to

$$\frac{T_{mb} - T_w}{T_{cl} - T_w} = \frac{\int u / u_{cl} \cdot (T - T_w) / (T_{cl} - T_w) . dA}{\int u / u_{cl} . dA} \quad (7)$$



and Reynolds analogy gives

$$\frac{T_{mb} - T_w}{T_{cl} - T_w} = \frac{\int (u / u_{cl})^2 \cdot dA}{\int u / u_{cl} \cdot dA} \quad (8)$$

An other option is to take temperature measurements at two locations which have the same profile factor. The two equal values of  $\phi$  cancel out in equation (4). An example would be to use centreline thermocouples in a long ribbed passage which has achieved developed conditions. The temperature profile options are summarised in Table 2.

Option	Example	Comments
Profile measured directly at plane	None	Most accurate
Profile factor inferred from velocity measurement and Reynolds analogy.	Rib results present papers	
Profile shape estimated from insight into the flow field. Eg. the flow can be well mixed to the extent that the profile can safely assumed to be uniform.	Impingement work of Son et al. (2001)	Well mixed profile at exit to array.
Analysis applied between planes for which the profile is identical.	Developed flow in ribbed passage.	

Table 2 Summary of Profile factor evaluation options.

### 1. Analysis using average temperature and the initial temperature method.

The above integral was evaluated at plane 2 (Figure 4) as 0.85 over a grid of 2500 measurement points. This was very close to the average dimensionless temperature difference determined at the 9 thermocouple measurement sites. The latter was again determined from the velocity measurements using Reynolds analogy and was found to 0.84. In other words, the average gas temperature at plane 2 was an excellent approximation to the mixed bulk temperature.

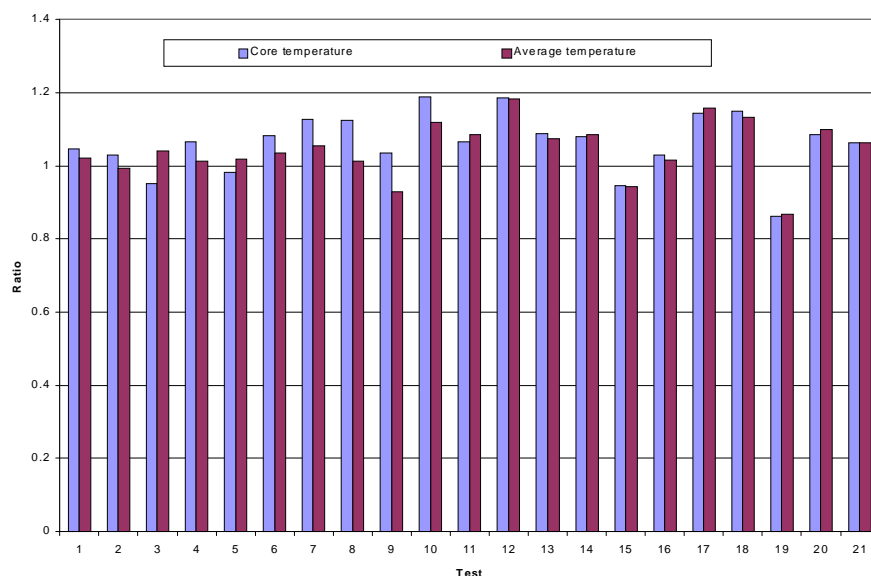


Figure 8 Ratio of heat transfer coefficient determined using the new technique (A) to h determined from a conventional liquid crystal heat transfer experiment.

For this reason, the profile factor was set to unity at the inlet and the average temperature used as the upstream experimental temperature in equation (4). All of the rib configurations tested act to stir the flow and the exit temperature profiles were found to be sufficiently flat

that the set average temperature was insensitive to the set of thermocouples. The exit gas temperature profile was also set to unity. The heat transfer coefficient enhancement values were then compared to data determined using more conventional liquid crystal heat transient experiments and the ratio between the two values compared for the rib tests. The results are shown in Figure 8.

## 2. Analysis using core representative temperatures and the initial temperature method.

In some situations, it will not be possible to determine an average temperature representative of the mixed bulk temperature. The data was reprocessed using the 3 core thermocouples (C, F and G) in Figure 2 at plane 2. A profile factor of 0.85 was used in the analysis and results (divided by the normal liquid crystal data) are presented in Figure 8.

### Uncertainty analysis: A. Initial temperature distribution method

Uncertainty analysis shows that the fractional uncertainty in  $h$  (assuming that uncertainties in  $A$ ,  $m$  and  $c_p$  are negligible), is

$$\frac{\sigma_h}{h} = \frac{1}{NTU} \sqrt{\left\{ \frac{\sigma_{\phi_i}}{\phi_i} \right\}^2 + \left\{ \frac{\sigma_{\phi_{i-1}}}{\phi_{i-1}} \right\}^2 + \left\{ \frac{\sigma_{T_{exo-W_i}}}{T_{exp_i} - T_W} \right\}^2 + \left\{ \frac{\sigma_{T_{exp-W_{i-1}}}}{T_{exp_{i-1}} - T_W} \right\}^2} \quad (9)$$

where  $\sigma$  denotes the standard deviation in a parameter. Recall that  $\phi$  is a factor that characterises the temperature profile. As defined above, it is the quotient of (mixed bulk temperature - wall temperature) divided by (measured gas temperature minus wall temperature). The ratio  $\sigma_{\phi_i}/\phi_i$  is the fractional uncertainty in this temperature profile parameter at the plane  $i$ . The equation for  $\sigma_h/h$  shows that the accuracy in the measured  $h$  is reduced if the gas measurement planes are too close together since the NTU between planes reduces. Figure 9 shows the calculated uncertainty for different uncertainty levels in the temperature profile factors. The legend indicates the permutation of inlet and outlet profile factor (and respective uncertainties) used. Unsurprisingly, the accuracy reduces as the latter uncertainties increase. The calculations include a contribution from uncertainty in the measured experimental temperatures. The gas temperatures were all measured with a data logger fitted to a PC that included a multiplexed amplifier and cold junction compensation. Since the same board is used to measure the gas temperature, the uncertainty in the gas temperature differences from the starting temperature is considerably lower than uncertainty in the absolute temperature. A level of 0.3°C for the standard error in temperature difference for both planes was used in Figure 9. The exit plane temperature difference is fixed by the inlet plane temperature difference and the NTU between planes so the above equation becomes

$$\frac{\sigma_h}{h} = \frac{1}{NTU} \sqrt{\left\{ \left\{ \frac{\sigma_{\phi_i}}{\phi_i} \right\}^2 + \left\{ \frac{\sigma_{\phi_{i-1}}}{\phi_{i-1}} \right\}^2 + \left\{ \frac{\sigma_{T_{exo-W_i}}}{T_{exp_i} - T_W} \right\}^2 + \left\{ \frac{\sigma_{T_{exp-W_{i-1}}}}{\frac{\phi_{i-1}}{\phi_i} (T_{exp_i} - T_W) \exp(-NTU)} \right\}^2 \right\}} \quad (10)$$

From the chart it is clear that the measurement planes need to be sufficiently far apart to achieve an acceptable accuracy. The uncertainty starts to rise as the NTU goes beyond 2 (dependent on the parameters) since the last term in equation (10) starts to dominate. Physically, the NTU needs to be sufficiently large between planes to ensure the gas temperature difference is measurable, but not so large that the exit plane temperature is the same as the wall temperature.

The NTU term in the denominator of equation (10) is responsible for the initial reduction of uncertainty with NTU. It is pleasing to note that the uncertainty in  $h$  can be less than the uncertainty in the profile factors.

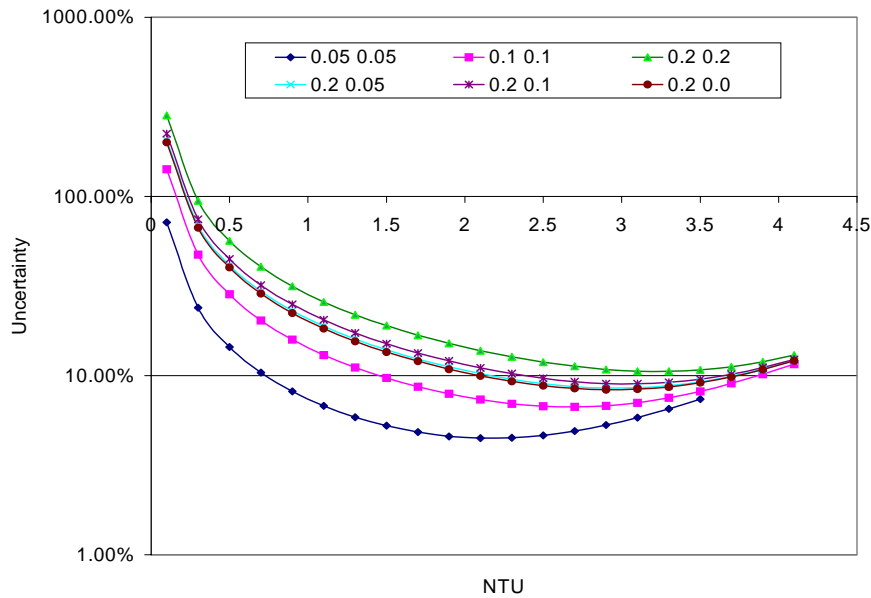
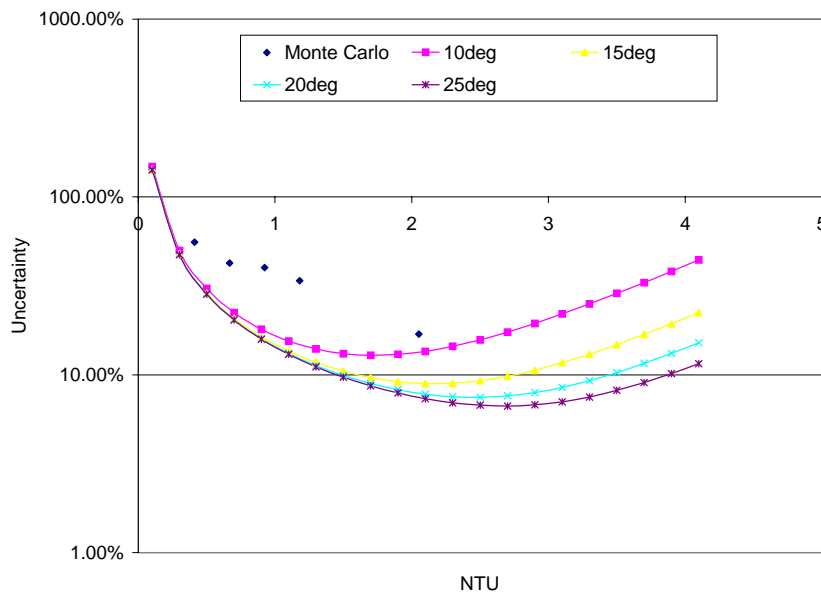


Figure 9 Calculated uncertainty in  $h$  as a function of the Number of Transfer Units between planes for different shape factor uncertainty levels. The uncertainty in both gas temperature minus initial temperature is set at  $0.3^{\circ}\text{C}$  and the inlet gas to starting temperature difference is  $40^{\circ}\text{C}$ .



**Figure 10** The diamond symbols show the calculated uncertainty in  $h$  as a function of the Number of Transfer Units between planes for method B. The uncertainty in both gas temperature minus initial temperature is set at  $0.3^{\circ}\text{C}$  and the uncertainty in the profile factors is 10%. The legend indicates the initial difference between the upstream gas temp and the model.

### Uncertainty analysis: B. Gas temperature variation with time method

A Monte-Carlo technique was used to determine the uncertainty in the  $h$  measurement for the second method and the results for different NTUs are plotted in Figure 10. The uncertainty shows the same trends as before and has approximately the same level.

## Conclusions

A new way of processing gas temperature data from apparatus used for transient liquid crystal experiment has been described and evaluated. The data resolution is reduced, relative to a liquid crystal experiment, but the method offers advantages in situations where processing time is restricted or optical access is not possible. The experimental uncertainty has been evaluated and acceptable accuracy is achieved when there is sufficient temperature change between measurement planes. The new method offers the potential to perform internal cooling experiments in metallic models at with higher heat flux.

## Acknowledgements

This work has been carried out with the support of Rolls-Royce plc, DERA, DTI CARAD and MOD ARP26c. The continuance of the support is greatly appreciated by the authors as is permission to publish the work. The authors thank the Croucher Foundation who provided Calvin Tsang with a scholarship. The authors are grateful to Mr Pat Timms who manufactured the experimental apparatus.

## Nomenclature

A	area, m <sup>2</sup>
c <sub>p</sub>	specific heat at constant pressure, Jkg <sup>-1</sup> K <sup>-1</sup>
h	heat transfer coefficient, Wm <sup>-2</sup> K <sup>-1</sup>
m	mass flow rate, kgs <sup>-1</sup>
NTU	number of transfer units
q	heat flux, Wm <sup>-2</sup>
T	temperature, °C

### Subscripts

exp	measured temperature
exp-W	difference between measured and starting wall temperatures
i	plane number
w	initial wall

### Greek

φ	temperature profile factor
ρ	density, kgm <sup>-3</sup>

## References

- Azad, G.S, Huang, Y, Han, J-C, "Impingement heat transfer on pinned surfaces using a transient liquid crystal technique," 8<sup>th</sup> International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Hawaii, USA
- Baughn, J. W., Ireland, P. T., Jones, T.V. and Sanei, M., 1989 "A comparison of the transient and heated coating methods for the measurement of local heat transfer coefficients on a pin-fin." ASME Journal of Heat Transfer, vol.111, pp.877-881.
- Clifford, R.J., Jones T.V and Dunne, S. D. 1983, "Techniques for obtaining detailed heat transfer coefficient measurements within gas turbine blade and vane cooling passages," ASME paper 83-GT-58.
- Chyu, M.K., Ding, H., Down, J.P. Sutendael, A.V. and Soechting, F.O., 1997, "Determination of local heat transfer coefficient based on bulk temperature using a transient liquid crystals technique," ASME paper 97-GT-489.
- Florschuetz, L.W., Metzger, D.E. and Truman, C.R., 1981, "Jet Array Impingement with Crossflow Correlation of Streamwise Resolved Flow and Heat Transfer Distributions," NASA Contractor report no. 3373.
- Gillespie, D.R.H., Ireland, P.T. and Dailey, 2000, "Detailed flow and heat transfer coefficient measurements in a large scale model of a novel internal cooling geometry employing a lattice of orthogonal intersecting channels" presented at IGTI ASME conference, Munich.
- Gillespie, D.R.H., Byerley, A.R., Ireland, P.T., Wang, Z., Jones, T.V. and Kohler, S.T., 1994. "Detailed measurements of local heat transfer coefficient in the entrance to normal and inclined film cooling holes, Journal of Turbomachinery, Vol. 118, pp. 285-290

Hippensteele, S. A. and Russell, L. M., 1984."Use of a liquid-crystal, heater-element composite for quantitative, high-resolution heat transfer coefficients on a turbine airfoil, including turbulence and surface roughness effects." NASA TM 87355.

Hwang J J and Lui C-C Lui,. 1999," Detailed heat transfer measurements in straight and turned trapezoidal ducts with pin-fin arrays," Int. J. Heat Mass Transfer, vol 42, pp.4005-4016.

Ireland, P. T. and Jones, T.V, 2000, Liquid crystal measurements of heat transfer and surface shear stress", Measurement Science and Technology, vol. 11, Number 7, July 2000 Institute of Physics.

Mee, D.J., Ireland, P.T. and Bather, S., 1999,"Measurement Of The Temperature Field Downstream Of Simulated Leading-Edge Film-Cooling Holes, *Experiments in Fluids*, .vol. 27(3) pp. 262-272.

Metzger D.E. and Larson, D.E. 1986, "Use of melting point surface coatings for local convection heat transfer measurements in rectangular channel flows with 90-deg turns", J. Heat Transfer, vol.108, pp.48-54.

Smith, E.M, 1996," Thermal Design of Heat Exchangers, " Wiley.

Son, C, Gillespie, D, Ireland, P and Dailey, G.M., 2001, "Heat Transfer and Flow Characteristics of an Engine Representative Impingement Cooling System, Journal of Turbomachinery, vol. 123, pp 154-160

Tsang, C.L., Gillespie, D.R.H., Ireland, P.T. and Dailey, G.M., 2001,"Detailed performance assessment of 16% blockage interrupted ribs at 60degree inclination in a square section turbine blade cooling passage,"

Tsang, C.L., Gillespie, D.R.H., Ireland, P.T. and Dailey, G.M., 2000," Detailed thermal analysis of transient heat transfer experiments." 8<sup>th</sup> International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Hawaii, USA.

Wang Z., Ireland P. T., Kohler S. T. and Chew J.,1998, "Heat transfer measurements to a gas turbine cooling passage with inclined ribs," Journal of Turbomachinery, Vol 120, pp 63-69.

Wolfersdorf, J., Hoecker, R. and Hirsch, C., 1997,"Data reduction procedure for transient heat transfer measurements in long internal cooling channels," 2<sup>nd</sup> Int. Symposium on Turbulence, Heat and Mass Transfer.

Paper Number: 4

Name of Discussor: J. von Wolfersdorf, ABB Baden Switzerland

Question:

How did you measure the heat transfer at the high blockage ribs in your conventional liquid crystal technique?

Answer:

The ribs were made from copper and were fixed atop of the liquid crystal coating. The conductivity of the rib is sufficiently high that the analysis for a calorimeter on top of a semi-infinite substrate can be used to determine each rib heat transfer coefficient.

The method is described in Wang et al. 1996, Heat Transfer measurements to a gas turbine cooling passage with inclined ribs, Journal of Turbomachinery, Vol. 170, pp 63-69.

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